A SELF EXCITATION AND CONTROL SYSTEM FOR WIND TUNNEL DYNAMIC STABILITY MEASUREMENTS

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ABSTRACT

The design and development of a fast acting self-excitation and control system based on the principles of regenerative and negative feedback for phase resonance testing of aerodynamic models in wind tunnels is described. Simulation tests and analysis using a linearized model of the drive system were carried out to establish the satisfactory performance of the controller. A brief discussion about the successful wind tunnel tests carried out using this excitation control system for dynamic stability measurements is also presented.

1. Introduction

Assessment of the stability and control characteristics of a flight vehicle is usually based on the wind tunnel measurements of aerodynamic derivatives on a scaled model. The commonly used techniques for stability derivative measurements are free-oscillation, and forced oscillation methods[1,2]. However, a different approach to evaluate dynamic derivative from random response has also been recently validated[3]. The widely used forced oscillation technique employs a spring mounted model which is free to oscillate about an axia as a single degree of freedom system. The excitation force to oscillate the model is so controlled that the model is oscillated at its undamped natural frequency and at constant amplitude. Under such test conditions, the spring force is balanced by inertia force of the system, and the total system damping is balanced by the applied moment. Thus the applied moment is a direct measure of system damping under static i.e., "wing-off" conditions. During the blow down test, due to aerodynamic reaction on the model, the total system damping changes. The difference in the applied moment under 'wind off and 'wind tin' conditions ia thus a measure of aerodynamic damping, from which the dynamic stability derivative can further be evaluated (Appendix A).

During the blow down wind tunnel tests, large disturbances on the model due to turbulence in the air-flow are inevitable. Also due to aeroelastic effects the natural frequency of model oscillation could change. Under certain test condition, due to aero-elastic, temperature and model configuration effects, the system damping may become neutral or even negative. Under these varying dynamic parameter conditions, the phase resonance of the model must be maintained in order to satisfy the primary critieria on which the forced oscillation technique is derived (Appendix A). Since the tests are to be conducted in a blow down tunnel with short run times, it becomes essential to establish steady state oscillations rapidly to allow for satisfactory data acquisition time. Further the model safety considerations dictate that the transient overshoots in setting up the resonant oscillations be also limited. The quantitative specifications for such a control system would be as follows:

i) The frequency of oscillation should be within 3 to 100Hz (depending on the designed aerodynamic model).

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ii) The amplitude of oscillation should be adjustable within 1° to 4° of angular displacement

iii) The model starting from rest should stabilize at the present amplitude level of oscillation in less than 2 seconds

iv) Transient overshoot beyond set levels of oscillation should be less than 20%

v) The system should control oscillations to preset level in the presence of random load disturbances

iv) The system should be capable of operation under positive, neutral and negative damping conditions,

The present paper deals with the design and development of such a fast acting self excitation control system for wind tunnel testing of aerodynamic models. The performance of the control system was initially evaluated by simulating the mechanical drive system on an analogue computer, A brief discussion about actual wind tunnel tests carried out using this control system along with its performance is also presented.

2. Self Excitation and Control System

Figure 1 shows schematically the self excitation and control system used for the measurement of dynamic derivatives in wind tunnels[4]. The aerodynamic model is mounted on a pair of strain-gaged cross-flexures which provide the necessary elastic suspension. An electromechanical shaker excites the model through a linkage mechanism. The displacement transducer, power amplifier and the control system together form a closed loop system. The control system consists of a positive feedback loop which automatically excites the model at phase resonance under varying parameter conditions and a negative feedback loop which provides for quick stabilization against disturbing forces and also for pre-setting the amplitude of oscillations.

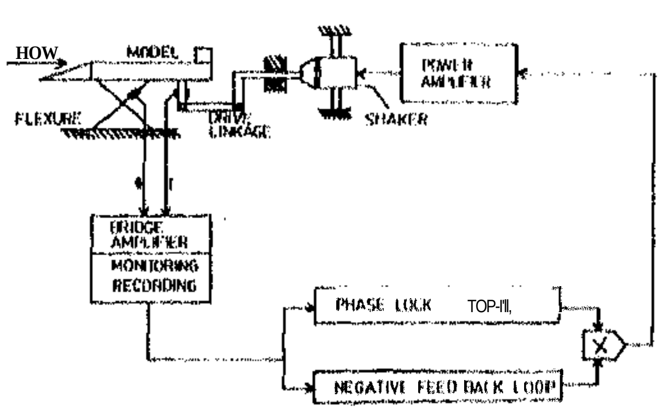

Fig. 1. Schematic of self-excitation and control system for phase resonance testing of aircraft models in wind tunnels

2.1 Phase lock loop

For the sustained oscillations at any amplitude level and phase angle around the closed loop system must be zero degrees. As the oscillations are to be built up starting from rest and large wide band disturbances are likely on the model it is necessary that the regenerative feedback circuit filter out these disturbances and also have a large amplitude output for easy and quick starting. Both these requirements are satisfied by the phase lock loop (PLL). The PLL also balances the 90° lag introduced by the test system driven at resonance by introducing 90° phase angle. This along with the 180° phase introduced by amplifiers of negative loop enables to maintain zero degree phase around the loop under test conditions. Also the sinusoidal high level constant amplitude output of PLL helps quick amplitude stabilization as discussed in the following section.

Figure 2 shows the schematic of the phase lock loop which consists of a phase detector, low pass filter and a voltage controlled oscillator. The phase comparator is essentially a four quadrant multiplier which mixes the VCO output with the input signal. The difference frequency ($f_R - f_0$) signal is low pass filtered and the averaged error signal e_m(avg) drives the VCO to keep the loop in lock.

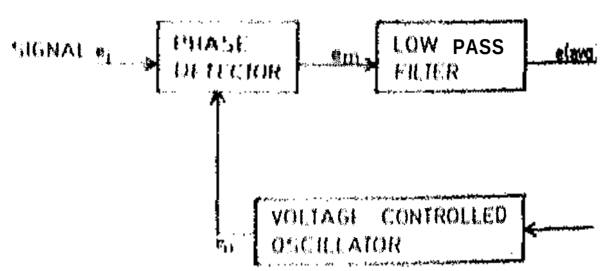

Fig. 2. Block diagram of phase lock loop

The mechanical drive system used to convert to and fro translatory motion of a shaker into an angular moment and the electromagnetic coupling of the shaker may introduce some additional phase shift which may result in a deviation of phase angle from 0° around the loop. This can be rectified by introducing an additional phase correcting reference signal in the error amplifier A2 of the PLL as shown in Fig.3. It was found to be possible to correct the phase deviation upto ±15° by this method.

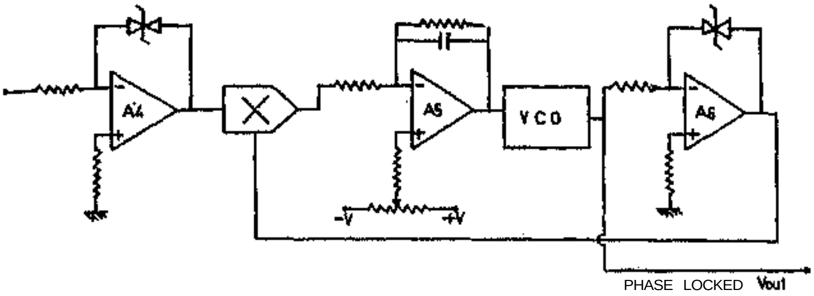

Fig.3. Phase Lock Loop-Circuit Diagram

As the phase locked output is power boosted and used directly to drive the electro-mechanical shaker any distortion in VCO's output waveform and in turn those of PLL's output will result in distortion of the input force. This makes a linear and highly stable voltage controlled oscillator a prerequisite. The VCO performance is optimized by matching the positive and negative feedbacks [7]. Fig.4 gives the circuit details of voltage controlled oscillator. The time constants of the two integrators A4 and A5 of the oscillators are so chosen as to cover the frequency range of interest[7].

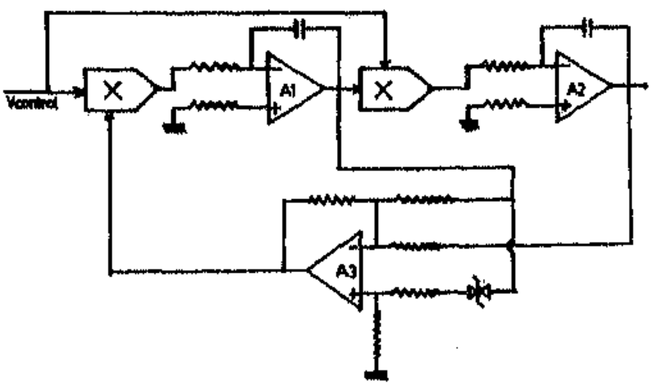

Fig.4. Voltage controlled oscillator-circuit diagram

2.2 Amplitude stabilizer

The stabilization action and the amplitude control in the presence of disturbing forces is obtained by using a multiplier as a gain controlling element in the negative feedback loop. The high amplitude sinusoidal putput of PLL ensures linear operation of the multiplier.

In the negative feedback loop, a dc signal proportional to the peak amplitude of oscillation (AB output of Fig.5) is compared with the preset amplitude level at every cycle. The error signal (A10 output) is low pass filtered with proper time constant (amplifier A12) and gain. This error signal modules the PLL output to control the drive signal to the shaker. The complete circuit details of the negative feedback loop are shown in Fig. 5.

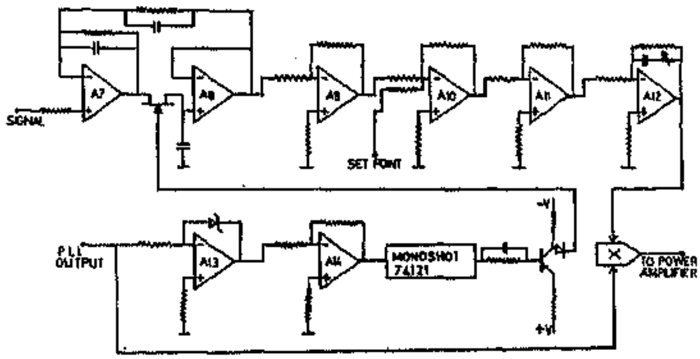

Fig.5. Negative feedback loop and sample command pulses - circuit diagram.

The number of amplifier stages in the negative feedback loop are so chosen that the, net phase around the complete closed system is zero degrees. The PLL output and the error signal mixed by the gain controlling multiplier, controls the amplidue of oscillation even in the presence of disturbing aerodynamic forces. When the amplitude of oscillation tends to decrease, the error signal automatically increases, thereby increasing the input to shaker, and consequently input force bringing the oscillation amplitude to deaired'level and vice versa. Under particular combinations of aero-elastic forces when the damping becomes negative, the amplitude of oscillation tends to increase rapidly. But any increase in amplitude beyond the set point causes the error signal to change polarity resulting in a 180° phase shift in the drive signal. The shaker drive now serves to oppose the model oscillation to bring it under control.

The quick stabilization action necessary as discussed above is obtained from the sample and hold circuit[7]. The sample command derived from PLL output, holds the positive peak. The transition time for change in the dc signal corresponding to peak is very small. And as the sampled value is up dated every cycle there is no averaging time involved in sensing the amplitude deviation. The timing sequence of the sample and hold action is shown in Fig.6.

The ripple in the controlled oscillations, which may arise due to low loop damping and various electronic components could be minimized by adjusting the zero of the overall function, which is equivalent to introducing a derivative control. The low frequency noise components rejection could bo achieved by tuning the low pass filter time constant in amplifier A12 in the negative feedback loop of Fig. 5.

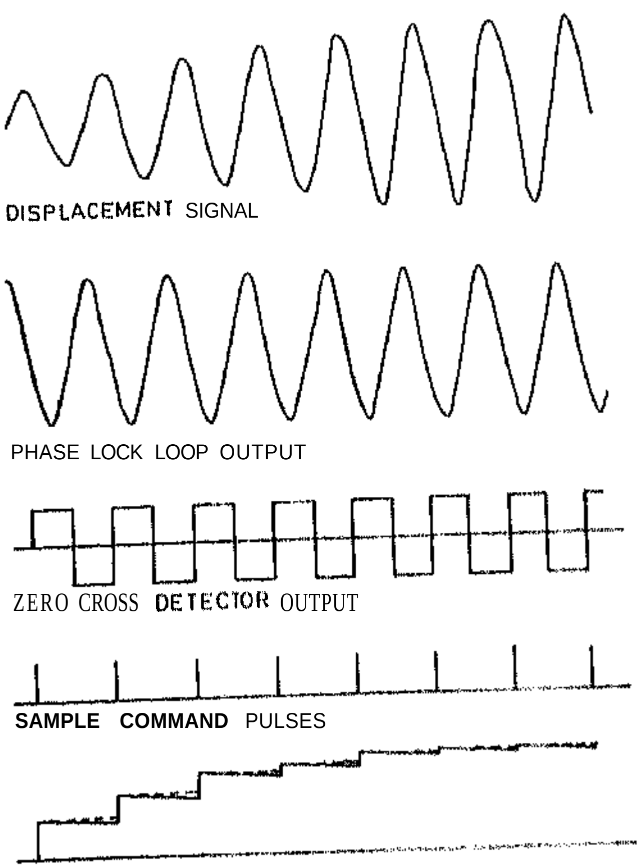

Fig.6. Timing sequence and output of sample and hold circuit for growing input. signal.

3. Analysis

For the purpose of analysis, the linkage drive mechanism, shaker and the power amplifier together with the aerodynamic model is modelled as a single degree of freedom system represented by the transfer function[8],

$$\frac{E_0(s)}{E_i(s)} = \frac{\omega_n^2}{s^2+2\xi\omega_n s+\omega_n^2} \qquad (1)$$

where ω_n is the natural frequency of the system and ξ in the damping ratio,

This system is excited by the developed control system to evaluate UN performance. The overall equivalent closed loop system is represented as shown in Fig. 7. The system output which is sinusoidal is represented by,

$$V(t) \quad V_\theta \frac{\omega_n}{s^2+\omega_n^2} \qquad (2)$$

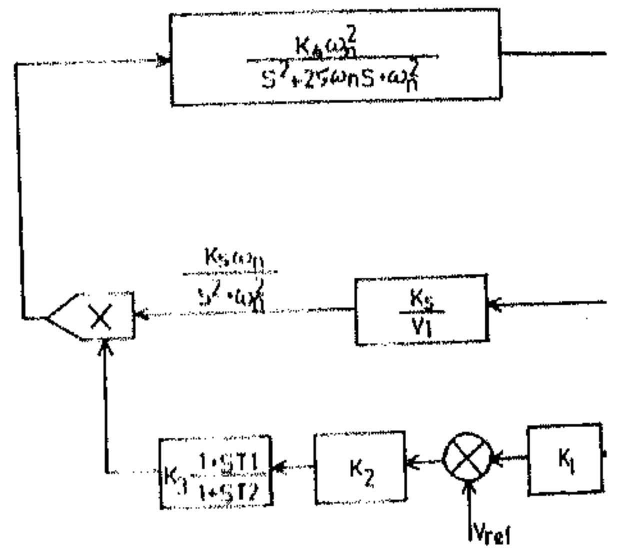

Fig. 7. Equivalent control system

As the PLL output is phase shifted by 90 with respect to input signal and of constant amplitude sinusoid independent of input signal level as a first order approximation it is assumed to be equal to $\omega_n/(s^2+\omega_n^2)$. The effects of the low pass filter time constant in the PLL has been ignored.

Neglecting the small ripple and droop in the sample and hold output (shown in Fig.6), from Fig. 5, we have,

Output of Difference Amplifier $= k_1(V_R - V_\theta)$ (3)

where,

V_R is the reference amplitude level
V_θ is the sampled peak value
k_1 is the gain

Then,

$$\text{LPF output} = k_1 k_2 k_3 \frac{1+T_1 s}{1+T_2 s}(V_R - V_\theta) \qquad (4)$$

where $k_3(1+T_1 s)/(1+T_2 s)$ is the lpf transfer function and k_2 is the gain of the amplifier.

Now from the equivalent system of Fig.7, we have

System output =

$$\frac{k_1 k_2 k_3}{10} \; \frac{1+T_1 s}{1+T_2 s} \; \frac{k_s \omega_n}{s^2+\omega_n^2}(V_R - V_\theta) \qquad (5)$$

The system output is now obtained from equations 1 and 5 as

$$V(s) = K \cdot \frac{\omega_n^2}{s^2+2\xi\omega_n s+\omega_n^2} \cdot \frac{1+T_1 s}{1+T_2 s} \cdot \frac{k_s\omega_n}{s^2+\omega_n^2}(V_R - V_\theta) \qquad (6)$$

where

$$K \text{ a } \frac{k_1 k_2 k_3 k_s}{10}$$

Equating cqns. 2 and 5, and rearranging

$$\frac{V_\theta}{V_R}(s) = \frac{K(1+T_1 s)}{(1+T_2 s)(1+(2\xi/\omega_n)S+1/(\omega_n^2)s^2)+K(1+T_1 s)} \qquad (7)$$

The characteristic equation of the system is given by

$$(1+T_2 s)(1+\frac{2\xi}{\omega_n}s+\frac{1}{\omega_n^2}s^2)+K(1+T_1 s) = 0 \qquad (8)$$

An analogue simulated second order system for mechanical drive was initially used to study the control system performance (Appendix B). This simulated system was driven in closed loop and the equivalent block diagram is shown in Fig.7.

Numerical values of the parameters for the various blocks are the following:

ω_n = 200 Rad/sec
ξ = 0.005 (Typical)
$1+T_1 s$ a $1+0.02s$
$1+T_2 s = 1+508$

The characteristic equation given by eqn. (8) now reduces to

$$12.5s^3+25.25s^2+(50+0.02K-0.05\times10^{-4})10^4 s$$
$$+(1+K)\times10^4 = 0 \qquad (9)$$

The Routh's tabulation for the equation (9) of the overall system gives the limits of stability as[8]

$$-1 \leq K \leq 104.2 \qquad (10)$$

To evaluate the transient response for a step input signal, substitute for $V_R(s) = 1/s$ in eqn, (7) and rewriting after factorization we get

$$V_\theta(s) = \frac{5.28 \times 10^4 \times (1+0.02s)}{s(s+1.305)(s+0.3575+j202.61)(s+0.3575-j202.61)} \qquad (11)$$

From eqn, (11), the time response of the overall system for a step input is given by

$$V_\theta(t) = 0.986-0.959e^{-1.305t}+1.31e^{-0.3575t}$$
$$\text{SIN}[202t-103°] \qquad (12)$$

The equation does not show the steady sinusoidal response term because the time response $V_\theta(t)$ is derived for output of sample and hold circuit, which is dc level corresponding to the peak of the sinusoidal response. Since the damping values encountered in the dynamic stability measurements are very low, typically of the order of $\xi = 0.005$, the damped natural frequency, $\omega_d = \omega_n$, the undamped natural frequency. The time for maximum overshoot can be calculated from pole zero locations of the overall transfer function[8], and for present case we get

$$t_p = 0.017 \text{ seconds} \qquad (13)$$

Therefore, the maximum overshoot from eqn. (12) is obtained as

$$V_{\theta max}(t)\Big|_{t=tp} = 13.19\% \qquad (14)$$

Thus the estimated overshoot from the above linear analysis is 13% and is less than the maximum allowable level of 20%.

The above linear analysis and tests with simulated system which are discussed in next section were carried out to establish the performance of excitation system. This transient analysis does not lead to damping derivative measurements. The damping derivatives are to be evaluated from the measurements of applied moment for maintaining phase resonance under "wind off" and "wind on" conditions as discussed in Appendix A.

Tests and Results

The mechanical drive system was analogue simulated to initially evaluate the performance of the control system. The designed control system drives the analogue system in closed loop. The expected varying parameter conditions were created and disturbances introduced in closed loop to study effect on control system performance. It was observed that the system could initiate and stabilize the oscillations at the preset level within 1.5 seconds. The amplitude control was excellent even in the presence of wide band noise injected in the loop. The transient overshoot was 12.85% which agrees with the linear analysis (13%) carried out. The deviation in this could be due to the linearized model of PLL used in analysis. In the end, it was observed that the control system controls the amplitude of oscillation even when the system damping becomes neutral or negative by appropriately changing the sign of the applied moment. Interestingly, the negative damping conditions can easily be simulated as shown in Fig.10 (Appendix B). Thus the performance characteristics were established prior to actual wind tunnel testing, which is briefly discussed now.

The forced oscillation rig consists of a model mounted on a pair of strain-gaged cross flexures, in such a way as to be free to oscillate in pitch axis. The model, initially locked mechanically necessitated by the starting loads, is allowed to oscillate when the flow is established. After the amplitude of oscillation stabilizes, the angular displacement signal and input excitation moment signal are recorded for further analysis[4]. Similar data is recorded under "wind off" conditions.

The displacement signal recorded during a typical blow down test is shown in Fig.8. The Initial portion of the response when model is locked before the flow is established shows the nature of flow disturbances likely on the model. As seen in Fig. 8, the model stabilizes to the preset amplitude level of oscillation within 1.5 seconds, thus ample time is left for data measurement for further analysis. The transient overshoot was measured under various test conditions, and was found to be 10% maximum. The deviation in maximum overshoot from linear analysis (13%) is due to the additional aerodynamic damping during the wind on test conditions. The linear analysis results and simulated systems results provide the worst case conditions. The overall amplitude of oscillation throughout the test was maintained within 5% of reference level against large flow disturbances. During the "wind off" tests, since the disturbances much less, the amplitude control was far bet The applied moments and angular displacem signals were further used for evaluating the mic. derivatives.

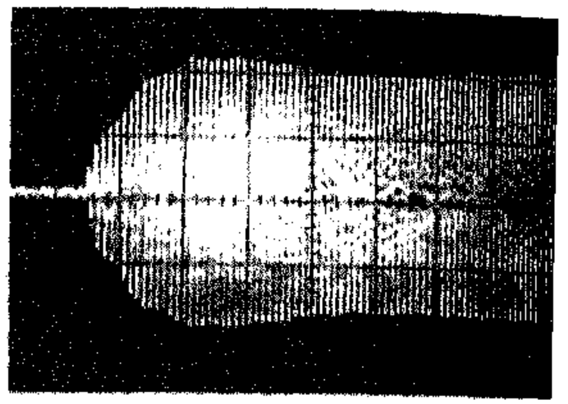

Fig.8. Typical displacement signal recorded during the wind tunnel test.

Conclusions

A fast acting self excitation and contro sys- tem with required specifications was desig d and developed which facilitates the phase n o- mance testing of wind tunnel aerodynamic m del under conditions of varying parameters and dis- turbances. The analytical theory presented included an approximation for the PLL mo l. The tests made with simulated system for mecha- nical drive have indicated that system woul function even if the system damping becom neutral or negative. The actual experi ents for dynamic stability measurements in the ri- sonic wind tunnel have shown that the cont i system meets all the specifications and pe forms satisfactory even in the presence of mode- rate amount of turbulance in the air flow.

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APPENDIX A

In the forced oscillation technique, the model is mounted on a pair of gaged cross flexures which provides necessary elastic suspension. The model, free to oscillate about an axis, as a single degree of freedom system, is forced sinusoidally and the amplitude of oscillation is controlled. The measurement of frequency of oscillation, amplitude of model displacement and exciting moment at 'wind off' (static) and 'wind on' conditions provide the necessary information for computing the aerodynamic derivatives,

The forced Oscillation motion of the model is governed by the following equation:

$$I\ddot{\theta} + D\dot{\theta} + K\theta = m_\theta \theta + m_{\dot{\theta}} \dot{\theta} + T$$

where 9 ia the angular displacement and m_θ and $m_{\dot{\theta}}$ are aerodynamic stiffness and damping derivatives. Following the notation in Kef. (4), we can derive the following suitable expression from above governing equation for aerodynamic derivatives in terms of change in applied moment. For sinusoidal excitation force, when model is oscillated at natural frequency and constant amplitude we get,

$$C_{m_q} + C_{m_\alpha} = \frac{2V_\infty}{q_\infty S d^2}\left[\frac{\overline{T}_0}{\omega_n \theta_0} - \frac{\overline{T}_1}{\omega_1 \theta_1}\right] \tag{15}$$

where V_∞ is free stream velocity, S ia model reference area, d is model reference length, q_∞ is free stream dynamic pressure, CO is frequency of oscillation and 9 is amplitude of oscillation. The index '0' refers to 'wind off conditions and '1' refers to 'wind on' conditions.

Thus from the measurement of ω_n, 0 and difference in average applied moment $(\overline{T}_0 - \overline{T}_1)$ the aerodynamic damping derivative can be evaluated.

APPENDIX B

In order to atudy the effect of various parameters like changes in natural frequency of oscillation, damping factor, and amplitude disturbances and to verify the amplitude control under conditions of neutral and negative damping, a simulated system was used to test the performance of the control system. For this purpose the transfer function of equation (1) representing the single degree of freedom system was electronically simulated as showin in Fig. 9.

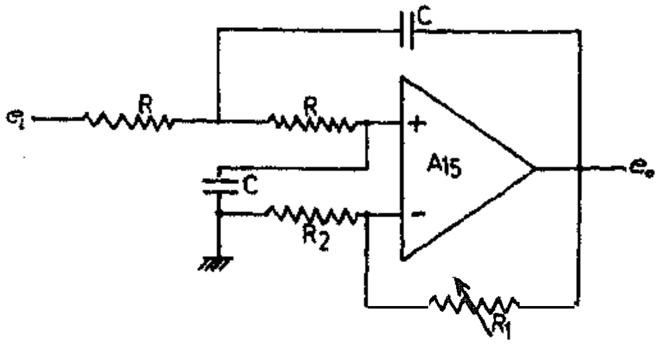

Fig. 9. Simulated second order test system.

This has a transfer function given by

$$\frac{E_o(s)}{E_i(s)} = \frac{1+R1/R2}{1+RC(2-R1/R2)s+R^2C^2s^2} \tag{16}$$

From the above equation, we have

Natural frequency ω_n - 1/RC

Damping ratio ξ = 1 - R1/2R2

Thus by adjusting the RC time constant in Fig. 9» any natural frequency of oacillation can be obtained to study its effects. The damping ratio ξ can also be varied by changing the resistor R2. The equation for ξ shows clearly that negative damping conditions can also be simulated with this type of realisation of the transfer function.

The amplitude control effectiveness can be easily verified by introducing a wide band random noise in the simulated system.

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